BENTLY ROTOR DYNAMICS RESEARCH CORPORATION

Shaft CenterLINES

Fluid-generated Instabilities of Rotors

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1. Introduction

Bently Rotor Dynamics Research Corporation (BRDRC) has been conducting research on fluid flow-generated rotor instabilities in fluid handling rotating machines for over seven years. The results are reported in a series of publications (see References).

In this article, a summary of the results is presented in a simplified form. Noted here are the bottom line, qualitative descriptions of physical phenomena, and approximate analytical expressions describing these phenomena.

2. What is rotor instability?

The situation when the shaft of a rotating machine exhibits high lateral vibrations which are not correlated to unbalance or other external periodic forces or constant radial forces (such as gravity) is informally referred to as "rotor instability". These vibrations usually have subsynchronous frequencies.

A more formal definition is that the rotor exhibits ***self-excited vibrations*** induced by some internal mechanism which, in most cases, transfers rotational energy into shaft lateral vibrations. In particular, fluid flow (including gas and steam) can play the role of such an energy convertor.

3. Fluid circumferential flow — the main contributor to rotor instability

The physical phenomena can be briefly described as follows A shaft rotating in an enclosed fluid environment (such as in bearings, seals, or a stator/case) drags the fluid into rotative motion. Usually the fluid flow is three-dimensional (spatial). However, the circumferential component (most often shaft rotation generated) may appear quite significant, independently from the other components of fluid motion (radial and axial). The circumferential flow generates the dynamic effect of fluid dynamic ***rotating force*** which, in turn (or rather in a feedback loop), drags the rotor into lateral vibrations.

4. Main characteristics of circumferential flow in bearings and seals

For the purpose of analyzing and predicting rotor instability, the circumferential flow in bearings and seals is represented by two main factors: the fluid circumferential average velocity ratio, λ (lambda), and fluid film radial stiffness, K_B. Both of these factors are functions of many parameters. The most important among these parameters is, however, the shaft eccentricity, i.e. shaft displacement from concentric position closer to the wall of a bearing or seal. The radial stiffness K_B increases with shaft eccentricity (Fig. 1). The fluid average velocity ratio decreases (slightly) when the shaft operates at low eccentricity and decreases dramatically when eccentricity becomes higher (Fig. 2). At high eccentricity, the shaft approaches the stationary wall and "cuts off" the circumferential flow. While this flow situation mainly occurs in radial clearances, as in seals and bearings it may also occur in an axial clearance area (such as associated with balance pistons).

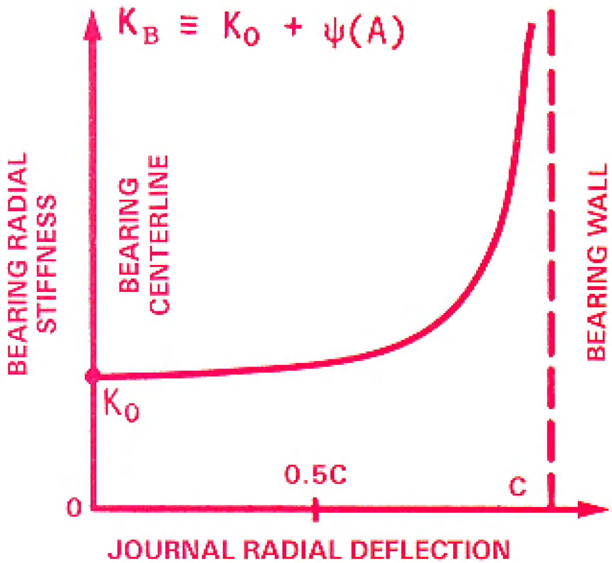

Figure 1
Fluid film radial stiffness as a function of shaft eccentricity; *c* is bearing radial clearance. Similar relationship holds true for seals.

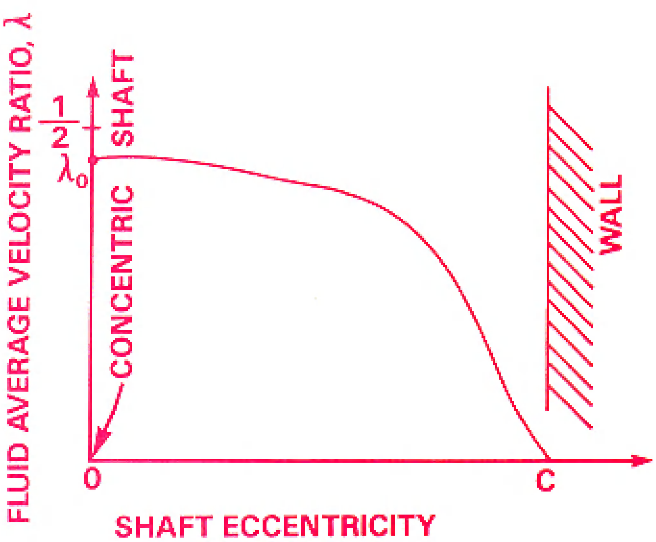

Figure 2
Typical fluid circumferential average velocity ratio as a function of shaft eccentricity inside a bearing or seal.

Fluid film radial stiffness K_B as a nonlinear function of shaft eccentricity represents a well recognized characteristic of bearings and seals. The fluid circumferential average velocity ratio λ is relatively new. It has been identified at BRDRC throughout extensive experimental testing [Ref. 1-16].

The more formal definition is as follows: Fluid circumferential average velocity $\lambda\Omega$ induced by shaft rotation with rotative speed Ω is the angular velocity at which the fluid flow-generated force rotates.

For any bearing or seal, both functions K_B and λ can be obtained experimentally [Ref. 4-16] and/or analytically from fluid dynamic theory equations [Ref. 23].

5. Whirl and whip

Fluid motion-induced self-excited lateral vibrations of shafts are known as "oil whirl/whip," "steam whip," "aerodynamic whip," or simply "rotor instability". All these vibrations actually belong in the same category. They are characterized by forward orbital motion (in the direction of shaft rotation) and subsynchronous frequency, which is either nearly constant (independent of rotative speed) or is a fraction of the rotative frequency, maintaining this fraction constant as the rotor speed varies, and when other parameters of the system are constant. This fraction is very ***close*** in value to the fluid circumferential average velocity ratio discussed above. The first situation is characteristic for "whip," the second for "whirl". The constant frequency of whip is usually ***close*** to the rotor natural frequency, most often corresponding to the first lateral mode (it could also be of a higher rotor mode) [Ref. 25].

There is a very smooth transition from whirl to whip with increasing rotative speed (Fig. 3a). Sometimes, when λ is close to 0.5 and the threshold of stability is higher than double the first natural frequency of the rotor first lateral

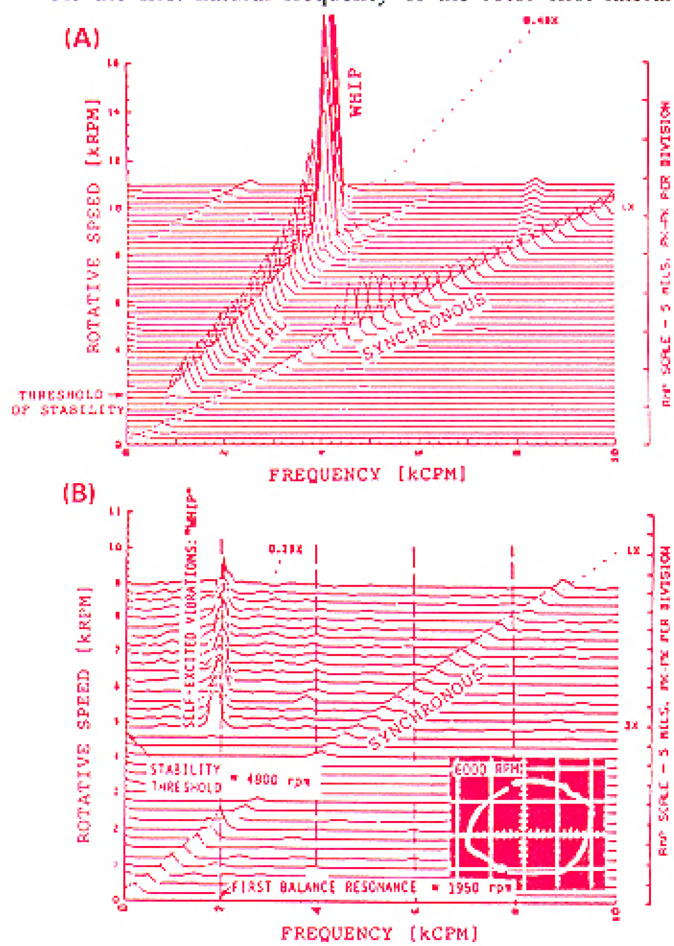

Figure 3
(a) Spectrum cascade of the vibrational response during startup of a rotor supported in one rigid and one oil-lubricated bearing. (b) Spectrum cascade of the vibrational response during startup of a rigidly supported rotor with a mid-span seal.

mode, the whip occurs without being preceded by whirl (Fig. 3b). At higher rotative speed, the whip may disappear and the rotor stabilizes. Sometimes whip "jumps" into a whirl of the second mode ("jump" relates to amplitude *and* frequency) (Fig. 4). Sometimes whip and whirl of the second mode exist simultaneously (Fig. 5).

6. Threshold of stability

Threshold of stability is the rotor speed at which self-excited vibrations start (Figs. 3, 4, and 5). Very informally speaking, at the threshold of stability, damping-type forces disappear, then become "negative". More precisely, the actual damping forces do not change at all, but there are emerging forces which act opposite to damping, first nullifying it, then becoming dominant.

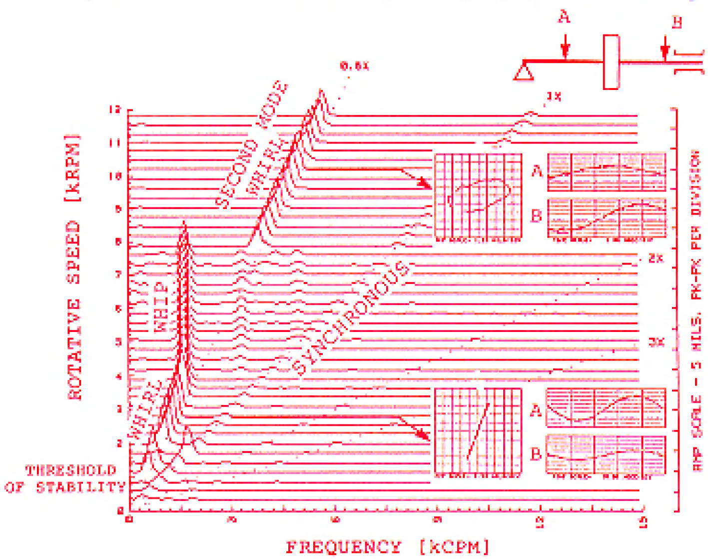

Figure 4

Spectrum cascade of the vibrational response during startup of a rotor supported in an oil-lubricated bearing indicating the "jump" of self-excited vibrations from whip to the second mode whirl. The sketch indicates where measurements were taken. The oscilloscope in orbital mode shows that shaft in whirl at lower rotative speeds vibrates in phase; at higher rotative speeds, shaft right-hand side is 180° out of phase from left-hand side.

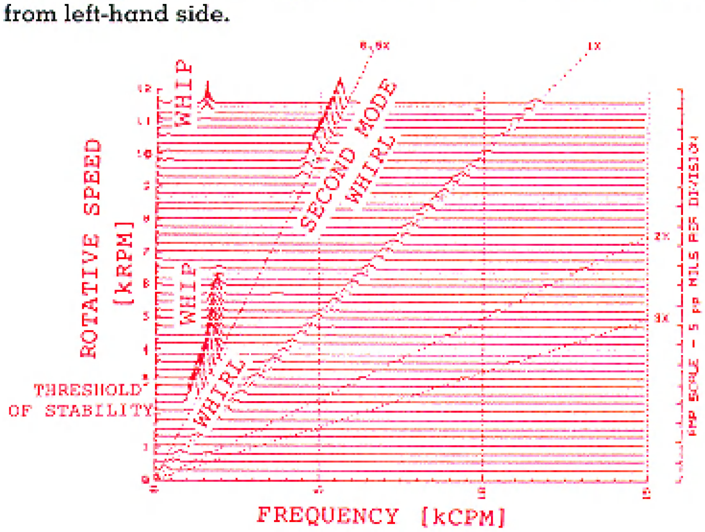

Figure 5

Spectrum cascade of the vibrational response of a rotor indicating coexistence of self-excited vibrations of the whip and second mode whirl types.

The "stable" rotor at rotative speeds below the threshold of stability rotates smoothly, and usually, due to residual unbalance, it exhibits some synchronous lateral vibrations (and possibly higher order harmonics). At the threshold of stability, the rotor starts vibrating with a subsynchronous frequency and increasing amplitude, finishing up in a limit cycle self-excited vibration of the whirl or whip category (Fig. 6). While this new subsynchronous component appears in the vibrational spectrum, the synchronous component does not exhibit any significant change.

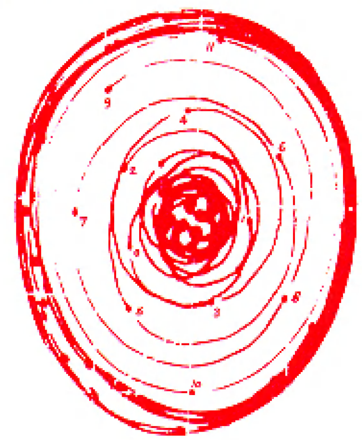

Figure 6

Oil whirl limit-cycle inception. Numbers on the orbit indicate consecutive rotations. Picture taken from oscilloscope display with exposure 1/3 second; shaft rotative speed was 3180 rpm.

The threshold of stability can be easily predicted by a ***linear*** mathematical model of the rotor system (such as represented by bearing or seal coefficients). However, the self-excited vibrations can only be described by a ***nonlinear*** model. The nonlinear terms in the model determine the values of vibration amplitudes, i.e., the size of the whirl and/or whip limit cycle orbits.

7. How to predict the threshold of stability

The threshold of stability does not depend exclusively on the bearing/seal characteristics; it is definitely an entire rotor/bearing/seal system property.

For a laterally isotropic rotor supported in one rigid bearing and one oil-lubricated bearing (Fig. 7), the threshold of stability, Ω_{ST}, can be approximated [Ref. 15] as:

$$\Omega_{ST} \approx \frac{1}{\lambda}\sqrt{\frac{K_0K_1+K_0K_2+K_1K_2}{M(K_0+K_2)}} = \frac{1}{\lambda}\sqrt{\frac{K_1+\dfrac{K_0K_2}{K_0+K_2}}{M}} \approx \left(\sqrt{K_1/M}\right)/\lambda \qquad (1)$$

where K_1 and K_2 are the partial stiffnesses of the rotor first lateral mode, M is the modal mass, K_0 is fluid film radial stiffness at zero eccentricity, and λ is the fluid circumferential average velocity ratio. Since the stiffnesses K_0 and K_2 act in series, their equivalent stiffness is smaller than either of them. Therefore, when either K_0 or K_2 is small, their equivalent stiffness becomes insignificant in comparison to K_1, and can be neglected. This yields the final approximation in Equation (1).

In the case of a rotor with a close to mid-span seal and two rigid supports (Fig. 8), the threshold of stability is relatively higher than (1):

$$\Omega_{ST} \approx \frac{1}{\lambda} \left(\sqrt{(K_1 + K_2)/M}\right) \qquad (2)$$

where K_1, K_2 and M are the corresponding partial stiffnesses and mass of the shaft first lateral mode.

In both of these simple cases, the threshold of stability is inversely proportional to the fluid circumferential average velocity ratio, λ, and to the square root of rotor modal mass. Since λ decreases with shaft eccentricity inside the bearing or seal, the threshold of stability ***increases*** with eccentricity. It is well known that when a high radial load is applied to the shaft (such as gravity for horizontal machines), it results in shaft displacement to higher eccentricity inside the bearings. This stabilizes the rotor: oil whirl or whip disappear.

Determined by the rotor actual mass, the modal mass M in the denominator of the threshold of stability is also significant: a heavier rotor usually results in ***lowering*** the threshold of stability. This sounds contradictory to the common beliefs. Note, however, that we are talking about the "mass" not the "weight" (as a force) and we do not specify whether the rotor is horizontal or vertical. It is assumed that the shaft rotates at the ***same*** original eccentricity, i.e., λ remains the same for the rotor with lower or higher mass.

Interesting also is the influence of rotor stiffness on the stability threshold. If at the rigid bearing side in the first model (Fig. 7) the rotor is stiff (K_1 higher), the threshold of stability increases. If it is stiff at the fluid lubricated bearing side (i.e., the rotor mass center is closer to the fluid bearing), the stability threshold decreases, i.e., the rotor is less stable (Figs. 9 and 10).

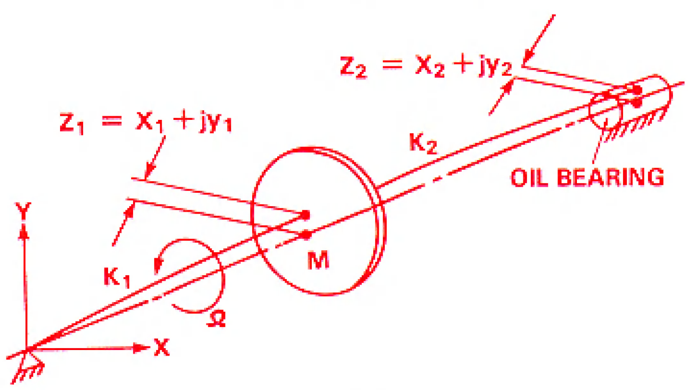

Figure 7
Model of a rotor supported in one rigid and one oil-lubricated bearing: x_1,x_2 and y_1,y_2 are shaft horizontal and vertical deflections respectively.

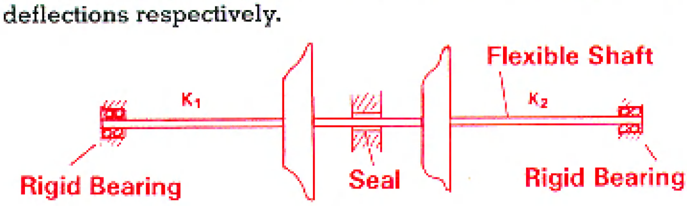

Figure 8
Model of a rotor/seal system.

8. What are self-excited vibrations?

Vibrations in mechanical systems are usually classified into three categories: free vibrations, forced vibrations, and self-excited vibrations (Fig. 11). The differences between them exist in the mechanism of ***energy*** supply to sustain the vibration.

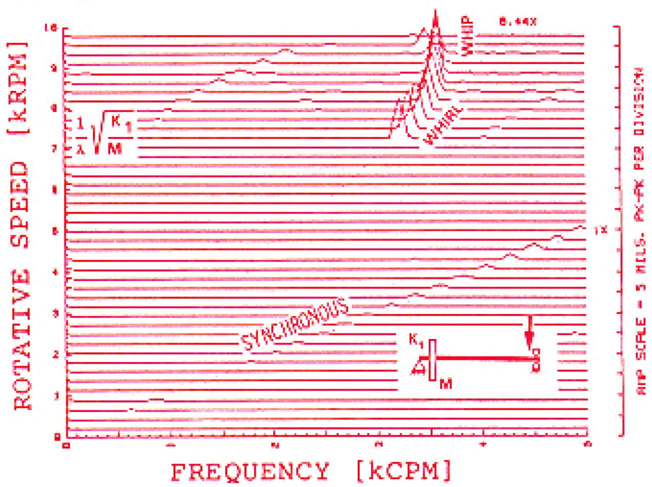

Figure 9
Spectrum cascade of the rotor vibrational response during startup. Heavy disk close to the rigid bearing. K_1 high, threshold of stability high.

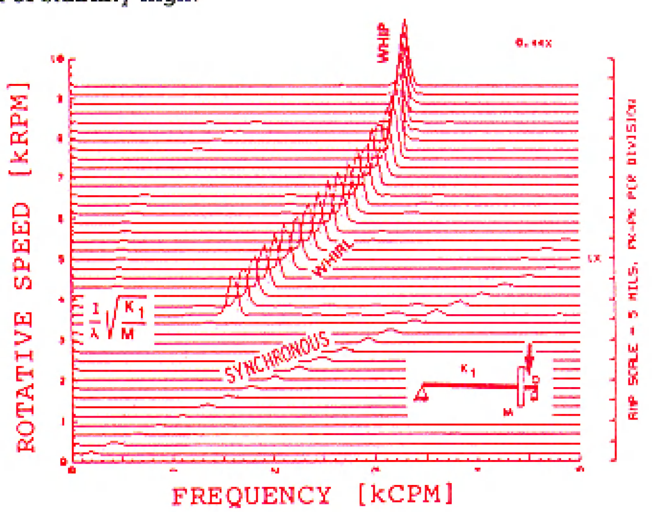

Figure 10
Spectrum cascade of the rotor vibrational response during startup. Disk close to the oil-lubricated bearing. K_1 low, threshold of stability low.

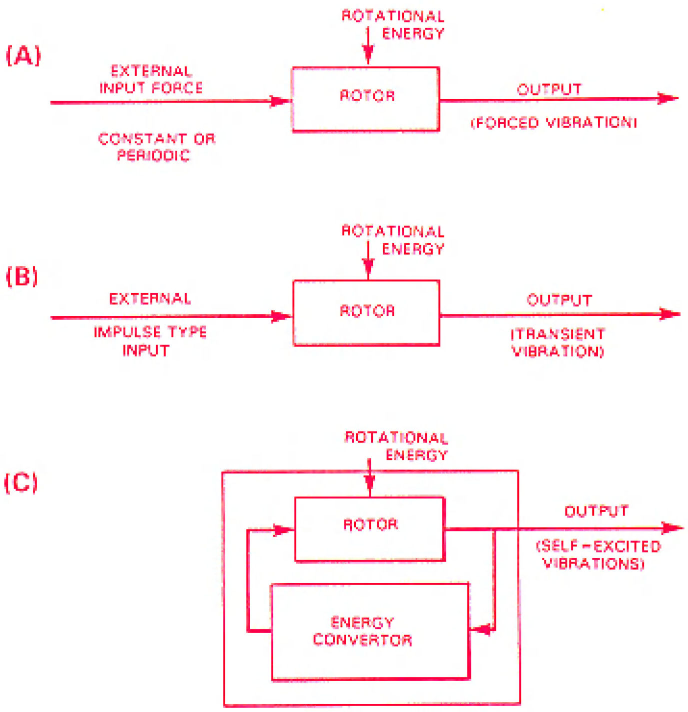

Figure 11
Free vibrations, forced vibrations, and self-excited vibrations in mechanical systems.

Free vibrations are of the transient type and they follow an instantaneous energy impulse. For lightly damped systems, the dominant frequency of free vibrations corresponds to the lowest natural frequency of the system.

In order to excite ***forced*** vibrations, the applied external force (thus the energy input) is usually periodic; the external forced vibration frequency is equal to that of the exciting force.

Note that both of these vibration types can exist in rotating or nonrotating structures. In general, the input force is independent from rotative motion. The resultive vibrational motion does not affect the input force.

Self-excited vibrations are induced by a constant force, sustained by a constant energy supply. The system has an internal energy transfer mechanism which delivers the energy in a periodic manner. The frequency at which the energy is provided usually corresponds to a natural frequency of the system.

A stable system responds to an impulse force with decaying vibrations; an unstable system responds with increasing vibration amplitude. Theoretically for unstable linear systems, this amplitude grows to infinity. Practically, the amplitude increase is accompanied by a significant increase of system nonlinearities which, in turn, slows down the amplitude growth. Most often, the final result is a "limit cycle" self-excited vibration. The amplitude of the self-excited vibration is, therefore, limited, and determined by nonlinearities in the system. The frequency of self-excited vibrations is very close to the linear system natural frequency at the threshold of stability, as it is usually insensitive to nonlinearities.

In rotor/bearing/seal systems, the constant energy input is represented by rotation. Fluid dynamic forces generated by shaft rotation in bearings and seals act as energy convertors from rotation to lateral motion and cause rotor self-excited vibrations of the whirl/whip type. Following all classical features of self-excitation in mechanical systems, these self-excited vibrations have a frequency close to one of the system natural frequencies, and the amplitude is determined by the system nonlinearities. The most dominant and significant among these nonlinearities are fluid film radial stiffness and fluid circumferential average velocity ratio as nonlinear functions of shaft eccentricity.

9. *Example: Natural frequencies and whirl/whip amplitudes of the rotor supported in one rigid and one oil-lubricated bearing*

In a very simplified form, the characteristic equation for natural frequencies and stability threshold of the linear rotor/bearing system illustrated in Fig. 7 is as follows (the formal analysis is given in [Ref. 15]):

$$(K_1 + K_2 - M\omega^2)\ [K_2 + K_0 + jD\,(\omega\text{-}\lambda\Omega)]\ -\ K_2^2\ =\ 0 \quad (3)$$

where ω is the complex eigenvalue, K_0 is the linear part of the fluid film radial stiffness, D is fluid film radial damping and Ω is rotative speed. This equation has three roots of which the real parts representing the system natural frequencies are approximately equal to the following values:

$$\omega\ \approx\ \pm\ \sqrt{(K_1 + K_2)/M} \quad (4)$$

$$\omega\ \approx\ \lambda\Omega \quad (5)$$

The natural frequency (4) corresponds to the rotor first lateral mode ("whip frequency"). The natural frequency (5) is due to the fluid/solid interaction and is called "whirl frequency."

The equation to calculate the frequency of the self-excited vibrations differs very little from equation (3):

$$(K_1 + K_2 - M\omega^2)\ [K_2 + K_0 + \Psi(A) + jD\,(\omega\text{-}\lambda\Omega)]$$
$$-\ K_2^2\ =\ 0 \quad (6)$$

where $\Psi = K_B - K_0$ is a nonlinear part of the fluid film radial stiffness versus eccentricity ($\Psi(0)=0$, Fig. 1) and A is the journal self-excited vibration amplitude ("a rotating eccentricity" in this case). The self-excited vibration frequencies will differ, therefore, very little from the natural frequencies (4) and (5). In this example, the fluid circumferential average velocity ratio λ is assumed to be constant.

The equation resulting from (6) for journal self-excited vibration amplitude estimation is as follows:

$$K_B\ \equiv\ K_0 + \Psi(A)\ =\ K_2\,\frac{M\omega^2 - K_1}{K_1 + K_2 - M\omega^2} \quad (7)$$

In equation (7), A and ω are journal amplitude and frequency of the self-excited vibration correspondingly.

The whirl amplitude can be estimated when the self-excited frequency is $\omega \approx \lambda\Omega$, obtained from the linear analysis. Equation (7) reads, therefore, (Fig. 12):

$$K_0+\Psi(A_{whirl}) = K_2 \frac{M\lambda^2\Omega^2 - K_1}{K_1+K_2-M\lambda^2\Omega^2} \quad (8)$$

Note that for a rotative speed lower than the threshold of stability, i.e., when

$$\Omega < \frac{1}{\lambda}\sqrt{\frac{K_1}{M}+\frac{K_0K_2}{M(K_0+K_2)}} \equiv \Omega_{ST}$$

the right-hand side function of (8) is smaller than $K_0 + \Psi$, and the two functions do not intersect (Fig. 13). This perfectly corresponds to the previously estimated stable rotor behavior below the threshold of stability.

For $\Omega > \Omega_{ST}$ both functions of Equation (7) cross, and at the intersection, yield the whirl or whip amplitude (Fig. 13).

When particular numerical values of the functions Ψ and λ are given, A_{whirl} can be calculated numerically. Usually the nonlinearity of λ at low and medium eccentricities introduces only a slight numerical adjustment, without modifying the qualitative picture of the phenomena.

The right-hand side function in Equation (8) increases to infinity when Ω approaches the value $\sqrt{(K_1+K_2)/M}/\lambda$. The latter value corresponds to the transition between "whirl" and "whip" natural frequencies (5) and (4) respectively.

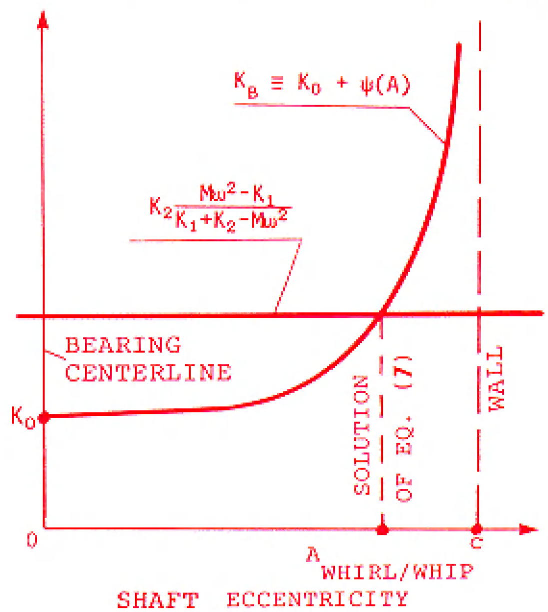

Figure 12

Estimation of whirl or whip amplitude based on Equation (7). $K_B = K_0 + \Psi(A)$ is fluid radial stiffness as a function of eccentricity; c is bearing clearance.

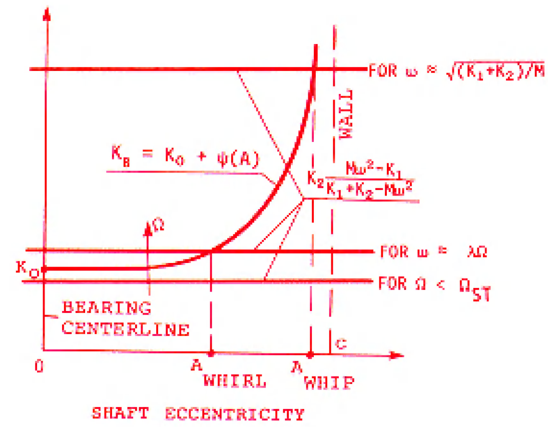

Figure 13

Estimation of the whirl and whip amplitudes (A) for various rotative speeds (Ω) and system natural frequencies ω, based on Equation (7).

From Equation (7), it follows also that when ω equals the whip frequency (4), the right hand side function (7) tends to infinity. Since the bearing radial stiffness K_B tends to infinity when the shaft approaches the bearing or seal wall, the journal whip amplitude also approaches the bearing wall. Thus the journal amplitude is limited by the bearing/seal clearance, c, and $A_{whip} \approx c$.

More important in the whip case is, however, to analyze the value of the rotor amplitude at mid-span, as the rotor vibrates at its resonant frequency. For both whirl and whip, this amplitude can be calculated from the following equation:

$$A_{rotor} = A_{journal} \frac{K_2}{\sqrt{(K_1+K_2-M\omega^2)^2+D_S^2\omega^2}} \quad (9)$$

with the relative phase between journal and rotor vibrations

$$\beta = \arctan \frac{D_S\omega}{M\omega^2-K_1-K_2} \quad (10)$$

where D_S is rotor modal damping.

For whirl, i.e. when $\omega \approx \lambda\Omega$:

$$A_{rotor} = A_{journal} \frac{K_2}{\sqrt{(K_1+K_2-M\lambda^2\Omega^2)^2+D_S^2\lambda^2\Omega^2}} \quad (11)$$

the amplitude A_{rotor} is less than $A_{journal}$, since $\Omega > \sqrt{K_1/M}/\lambda$.

The phase β is close to zero. The whirl mode has a "conical" shape (Fig. 14).

For whip, when $\omega \approx \sqrt{(K_1+K_2)/M}$ and $A_{journal} = c$, it follows from (9) and (10) that

$$A_{rotor} = c\left(\frac{K_2}{D_S}\sqrt{\frac{M}{K_1+K_2}}\right), \quad (12)$$

$$\beta = 90°, \quad (13)$$

thus, the rotor amplitude can be very high as it is controlled only by the rotor damping D_S. Note that the bearing damping D has no influence on the whip amplitude. The rotor vibrating at whip frequency is in resonance and, therefore, is in a highly dangerous condition. This condition is similar to prolonged rotor operation at the first balance resonance speed. But this time this jeopardizing condition is rotative speed independent. "Passing through," like the first balance resonance critical speed, is not possible. There exist, however, some other measures to correct the problem.

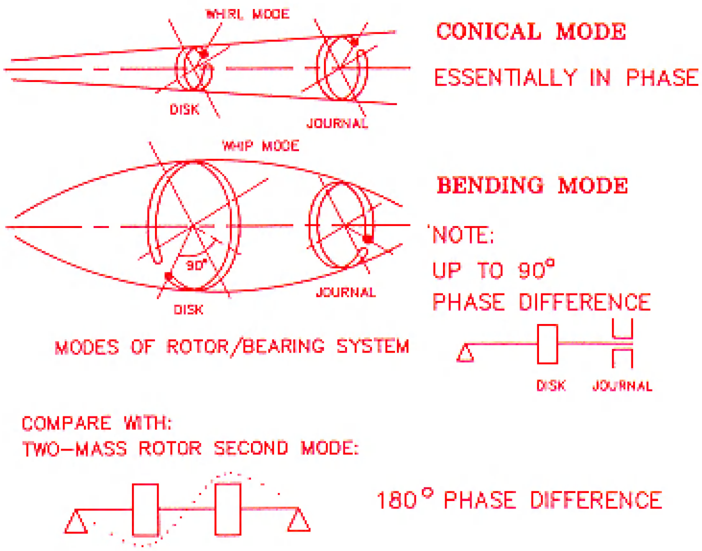

Figure 14
Whirl and whip modes of the rotor supported in one rigid and one oil-lubricated bearing.

10. Whirl and whip cures

Analyzing the threshold of stability (1) and (2) and whirl/whip amplitude evaluation (Equation (7) and Fig. 13), it is easily seen that there exist two major possibilities to improve the rotor stability, independently of rotor parameters: Decrease of λ and increase of K_B.

A decrease of the fluid circumferential average velocity ratio, λ, can be achieved by a perturbation of the circumferential flow regular pattern, or prevention of the circumferential flow occurrence in the bearing (or seal). In bearings, this can be successfully accomplished by noncircular geometries (e.g., elliptical, two or three lobe) and/or the presence of moving parts inside bearings (tilting pads, floating rings). In such bearings, the circumferential flow is significantly reduced in comparison to circular bearings.

In seals, a decrease of λ can be accomplished by injections of the fluid tangentially in the direction opposite to rotation in order to perturb the rotation-generated circumferential flow. This technique is known as "anti-swirl" control [Ref. 18,19] and is currently widely used in compressors, turbines, and pumps.

Moving the shaft to a higher eccentricity affects the bearing/seal radial stiffness K_B. At higher eccentricity, the radial stiffness is much larger and this improves rotor stability.

Higher eccentricities also lower the λ ratio which further improves stability. It requires an increase in the radial force applied to the shaft. It can be created either by a "friendly" misalignment, and/or by a heavier shaft in horizontal machines and/or by main flow distortion in fluid handling machines, so that the flow generates a significant radial force component.

An increase of fluid film radial stiffness, K_B, can also be achieved by increasing fluid pressure. Hydrostatic bearings (or more properly called "externally pressurized" bearings) are well recognized for their high stability features.

These cures which prevent rotor whirl and whip are illustrated in Figs. 15 to 18. The relationships are deduced from Equation (7).

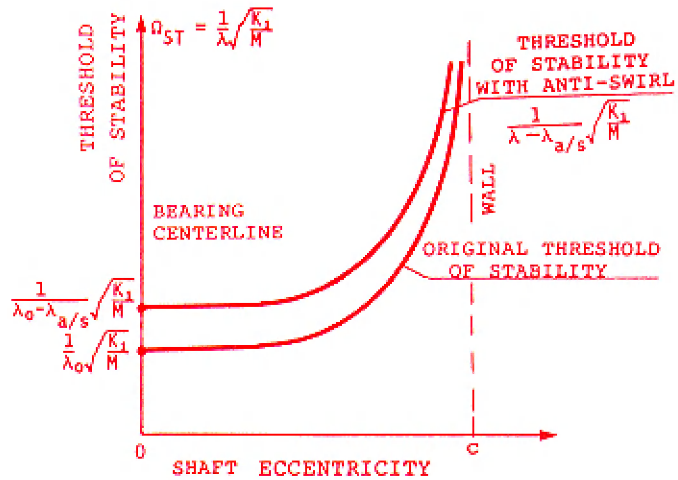

Figure 15
Instability cure: threshold of stability increases for shafts with fluid anti-swirl injections: $\lambda_{a/s}$ is the fluid average velocity ratio due to anti-swirl.

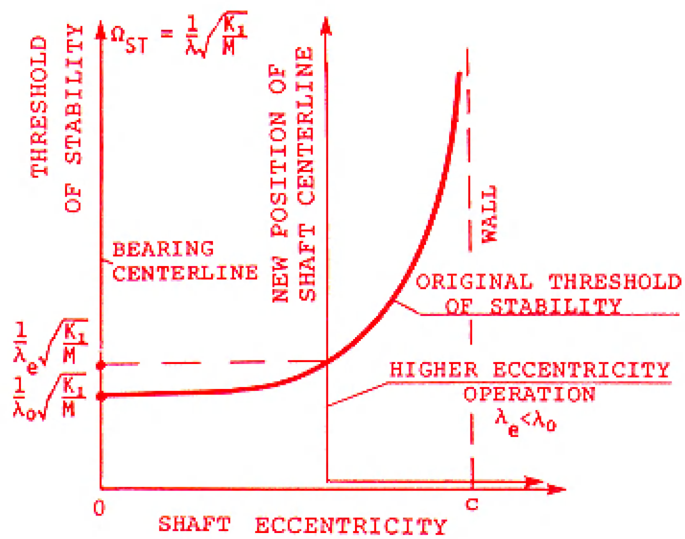

Figure 16
Instability cure: threshold of stability increases for shafts rotating at higher eccentricity: λ_e is the fluid average velocity ratio at higher eccentricity.

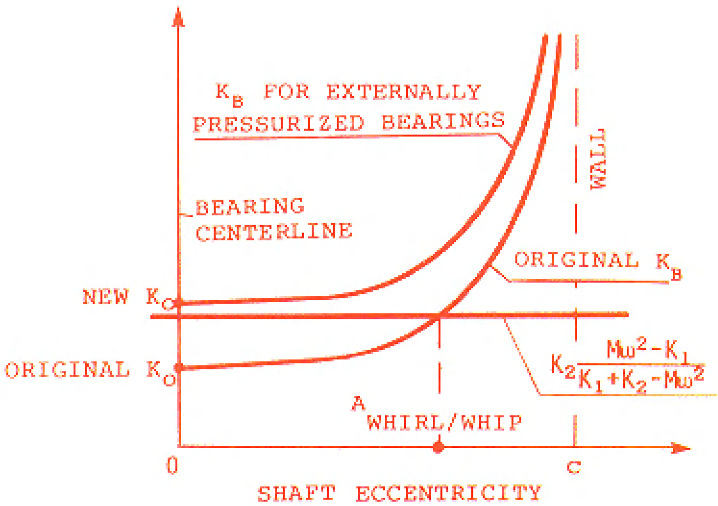

Figure 17
Improvements of rotor stability with externally pressurized bearings; K_B reaches a higher value than the threshold of stability.

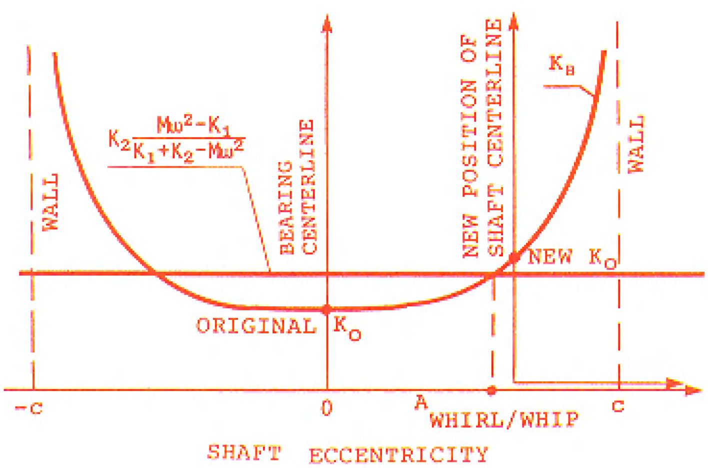

Figure 18
Instability cure: whirl/whip occurs when the shaft is concentric in the bearing or seal. Improvement of rotor stability is achieved when shaft operates at higher eccentricity: Function $K_2(M\omega^2-K_1)$ / $(K_1+K_2-M\omega^2)$ from equation (7) appears below the new K_0 and does not cross with K_B: No whirl/ whip exists.

11. Closing remarks

This article presents new results regarding flow-induced instabilities of rotor/bearing/seal systems. Experimentally identified models of fluid forces allow for better understanding of vibrational phenomena occurring in rotating machines and lead to rational prevention of undesirable flow-induced self-excited rotor vibrations.

Notation

A	Amplitude of rotor self-excited vibration of whirl or whip type.
$A_{journal}$	Amplitude of journal whirl or whip.
A_{rotor}	Amplitude of rotor mid-span whirl or whip.
A_{whip}	Amplitude of journal whip.
A_{whirl}	Amplitude of journal whirl.
c	Bearing or seal radial clearance.
D	Bearing or seal radial damping coefficient.
D_s	Rotor first lateral mode damping coefficient.
j	$\sqrt{-1}$
K_B	Bearing or seal fluid film radial stiffness, as function of shaft eccentricity.
K_0	Bearing or seal fluid film radial stiffness at zero eccentricity.
K_1, K_2	Rotor first lateral mode partial stiffnesses.
M	Rotor first lateral mode generalized mass.
t	Time.
$z=x+jy$	Rotor lateral deflection (x — horizontal, y — vertical).
β	Relative phase between rotor and journal self-excited vibrations.
λ	Fluid circumferential average velocity ratio.
λ_0	Fluid circumferential average velocity ratio for concentric shaft.
$\lambda_{a/s}$	Anti-swirl-related fluid circumferential average velocity ratio.
Ψ	Bearing or seal fluid film radial stiffness, a nonlinear function of shaft eccentricity.
ω	Rotor/bearing/seal system natural frequency, also frequency of self-excited vibrations.
Ω	Rotative speed.
Ω_{ST}	Threshold of stability.

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